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Research Article

Lightweight Investigation of Extended-Range Electric Vehicle Based on Collision Failure Using Numerical Simulation

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The total weight of Extended-Range Electric Vehicle (E-REV) is too heavy, which affects rear-end collision safety. Using numerical simulation, a lightweight method is designed to reduce E-REV body and key parts weight based on rear-end collision failure analysis. To calculate and optimize the performance of vehicle safety, the simulation model of E-REV rear-end collision safety is built by using finite element analysis. Drive battery pack lightweight design method is analyzed and the bending mode and torsional mode of E-REV before and after lightweight are compared to evaluate E-REV rear-end collision safety performance. The simulation results of optimized E-REV safety structure are verified by both numerical simulation and experimental investigation of the entire vehicle crash test.

1. Introduction

Fuel-efficient and new energy vehicles were being formulated with more severe engine emission standard and fuel consumption standard. But the design of fuel-efficient and new energy vehicles must be secondary development on the basis of traditional fuel vehicle and increase the parts such as power battery and drive motor. So, reducing the auto body weight is important to improve vehicle fuel economy. According to statistics, fuel consumption will be saved from 6% to 8% when 10 percent of vehicle weight is reduced. In other words, the fuel consumption per hundred kilometers will be cut down from 0.3 to 0.5 L and carbon dioxide emission will lessen from 8 to 10 g when the vehicle weight is reduced 100 kg. Therefore, reducing the vehicle weight is an effective measure for decreasing fuel consumption and emission pollution [1, 2].

There are two ways to achieve vehicle body structure lightweight [3]. One is by using light and high strength materials such as aluminum and high strength steel [4]; the other is by designing rational structure to make the parts thinner, hollow, and composite [5, 6].

The vehicles safety will be greatly affected as vehicle weight increases largely. So, the chassis performance must be adjusted and the crash performance must be analyzed. According to the car regulations, the new energy vehicles request not only the properties of good fuel economy and emissions but also the vehicle safety close to the traditional fuel cars.

Recently, the research on new energy vehicles has just started and the research results are limited. Most of automobile lightweight achievements mainly focus on traditional fuel vehicles. Therefore, it is necessary to study the analysis of vehicle lightweight technology based on crash safety.

In this paper, an Extended-Range Electric Vehicle (E-REV) will be analyzed to satisfy the lightweight design based on rear-crash safety.

2. The Basic Structure of E-REV

E-REV is a new pure electric vehicle with Range Extender (RE). When the battery power is enough, E-REV will move under the pure electric mode; if not enough, RE will work to charge the batteries or direct drive motor work [7, 8]. The rear floor structure of E-REV is shown in Figure 1.

The prototype vehicle is a mass-produced vehicle, so the structure of vehicle cannot be altered too much and can

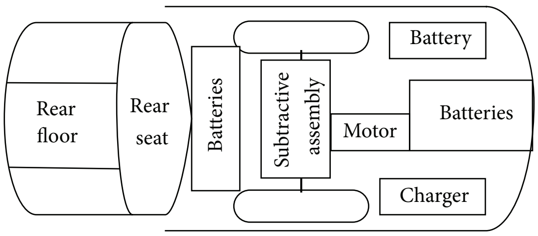


Figure 1: The rear floor structure of E-REV.

only transform in a small area. Compared with the prototype vehicle, the E-REV weight is increased about 270 kg.

The rear axle and rear floor structure of E-REV changed a lot (shown in Figure 1). In engine compartment, original engine is replaced by RE, and motor and poor fuel reduction are installed on rear axle. The battery pack 1 is added between the rear seats with car trunk. The battery pack 2 was placed on the location of the spare tire. The charger and the 12 v lead-acid battery were placed on both sides of car trunk; therefore, the drive motor and power battery boxes could bring great influence to the safety of vehicle rear-crash.

As E-REV before lightweight has passed standards GB/T 15083-2006 (seat and seat fastness equipment and pillow intension requirements and test method) [9], GB/T 19751-2005 (hybrid electric vehicles safety specification) [10], and GB 20072 (the requirements of fuel system safety in the event of rear-end collision for passenger car test) [11], the simulation results agreed well with the test results. Therefore, the same simulation method can be used to analyse lightweight design of E-REV.

3. The Finite Element Analysis of E-REV before Lightweight

According to standard GB 20072, the simulation model of E-REV is built and analyzed. The pre- and postprocessing of FEM choose OASYS and solver chooses LS-DYNA. The simulation parameters are based on the prototype vehicle test data. The material of vehicle body and fixed frame both choose steel, modulus of elasticity is 211 GPa, density is 7.85 g/cm^3, Poisson's ratio is 0.4, and yield limit is 340 MPa. The simulation model has 699924 elements, 718797 nodes, and 5799 welding spots [12].

3.1. The Finite Element Analysis Result of E-REV before Lightweight. Table 1 shows the finite element analysis results of E-REV body before lightweight with dormer, front, and back windshield glasses and front subframe. As shown in Table 1, the bending mode and torsional mode of the E-REV body meet with the design requirement. But the first torsional mode simulation result is close to the lower limits of target value. So it needs to improve.

The data of Table 2 come from prototype vehicle. As shown in Table 2, the BIW rigidity before lightweight is far greater than target value and satisfies the design requirements. Therefore, the NVH performance of E-REV after lightweight design must be similar to the prototype vehicle.

Table 1: The mode simulation results of E-REV before weight loss of main parts.

	First bending mode (Hz)	First torsional mode (Hz)
E-REV	44.5	38.3
Target value	≥35	

Table 2: The vehicle rigidity simulation results of E-REV before weight loss of main parts.

	BIW	BIW + front and back windshield + front subframe
Bending rigidity (N/mm)	28541	28302
Target value		≥8700
Torsional rigidity (Nm/°)	15232	19587
Target value	≥10500	≥16000

3.2. The Result of Rear-End Collision Simulation of E-REV before Lightweight. The simulation model is imported in the FEM software and is simulated. The processor chooses OASYS while solver chooses LS-DYNA. The result of rear-end collision simulation of E-REV is shown in Figure 2.

The simulation result shows that the right and left back stringer and subframe were greatly crumpled, and the crumple pattern is conductive to absorb the crash energy. The subframe moved backward and extruded fuel tank. The maximal stress of subframe is 244.6 MPa. The drive motor and subtractive assembly have a collision with fuel tank, but the collision force is not big. The maximal plastic deformation of fuel tank is 21 mm.

3.3. The Rear-End Collision Simulation Results of E-REV Battery Pack before Lightweight. As the E-REV rear bumper is impacted directly in the process of the rear-end collision test, the battery pack and its fixed support are greatly impacted and the battery pack fixed support needs to bear the high extruding strength. So the special structure must be designed to protect batteries when the battery pack can turn or move along the direction of the collision slope to provide enough space for absorbing crash energy in rear-crash test. At the same time, it can prevent the battery electrolyte leak as battery pack mutual extrusion or other potential risks. The simulation results are shown in Figure 3.

As shown in Figure 3, battery pack 1 can rotate some angle around the upper fixed point and give some deformation space to battery pack 2. The design scheme avoids two battery packs' mutual extruding and protects two battery packs.

The lower part of battery pack 1 is shocked largely and middle battery module deforms seriously. The result is present in Figure 4(a). The front left battery module bracket is severely crushed and severely deforms in rear-end collision simulation. The simulation result is present in Figure 4(b). The maximal deformation is over 30 mm. Battery pack 2 deforms apparently but the rear battery module deforms relatively smaller. Judging from the analysis results, it could

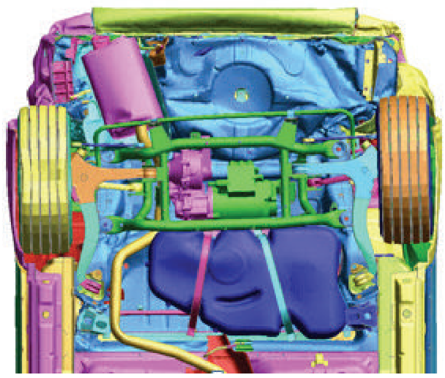


Figure 2: The rear-end collision simulation of E-REV.

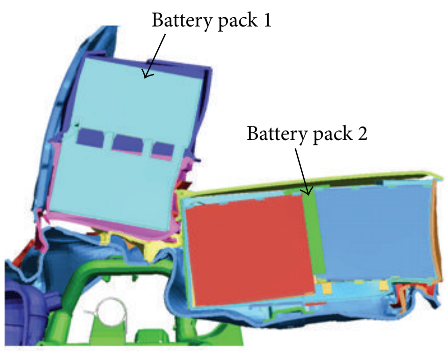


Figure 3: The impact simulation results of E-REV drive battery under rear-end collision test.

pass the requirement of standards GB 20072 and GB/T 19751. The deformation of two battery packs must be as small as possible. Otherwise, the risk of collision exists [13].

In rear-end collision test, the car trunk and low voltage battery on car trunk right stringer fixation were serious extrusion deformation; it may lead to battery electrode direct touch with vehicle body sheet metal. The potential risks require the special attention during lightweight design.

In conclusion, although the fuel tank does not leak and can pass the rear-end collision test, some risks still exist.

4. E-REV Lightweight Design Program

The lightweight design program of E-REV should consider not only the weight and vehicle performance but also the suitable price of parts. According to the optimization target of the lightweight design, all effective factors should be considered. So the problem of selecting lightweight design program is a complex multiobjective optimization problem. Look for the optimization values after considering all factors; then the lightweight design program could be gained.

4.1. Sensitivity Analysis of Vehicle Body Parts. In order to simplify the process, select the bending and torsional mode as the modes constraint condition and the vehicle body bending and torsional rigidity as rigidity constraint condition for optimization design. Then the sensitivity relationship between the thickness of the main vehicle body parts and torsional and bending sensitivity is obtained as shown in Table 3.

Table 3: The mode sensitivity of main parts thickness.

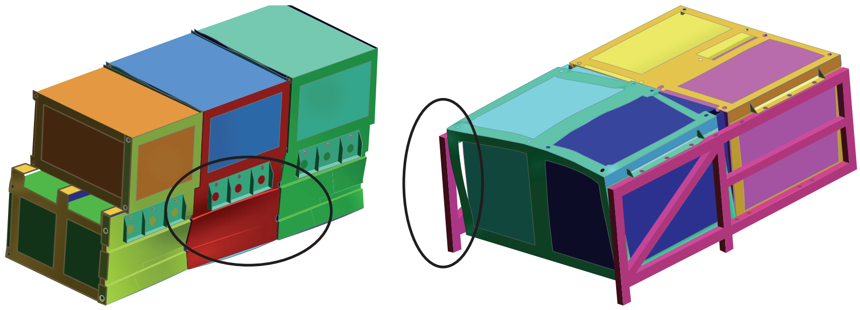
Variable number	Thickness (t/mm)	Torsional sensitivity ($Hz{\cdot}m^{-1}$)	Bending sensitivity ($Hz{\cdot}m^{-1}$)
02	1.5	−0.0065	−0.0057
03	1.5	−0.0017	−0.0023
07	1.5	−38.17	84.16
09	1.75	16.58	34.58
12	0.8	229.4	−843.7
15	0.8	−1563	−373.7
17	0.8	5198	4318
18	1.2	−687.3	1192
22	1.5	−1125	237.8
26	2.0	−1008	32.46
29	1.0	−754.1	398.7
35	2.0	−0.0018	0.0016
47	1.5	−0.0024	0.0031
55	1.5	−0.0026	0.0018
57	1.5	−0.0040	−0.0033

In Table 3, the sensitivity of thickness of number 15 (left stringer), number 17 (vehicle side panels), and number 22 (right stringer) is high for the first mode of the E-REV body. From the prototype vehicle test data, the vehicle front floor, middle floor, rear floor, rear stringer, threshold, and reinforced plate of the central channel constitute the vehicle basic torsional loading area; the area and its reinforced lateral confining structure components are effective to improve the vehicle first torsional mode. Therefore, it is important to avoid weak parts which are sensitive to the vehicle first mode and the main lightweight design targets should focus on parts which are insensitive to vehicle mode.

4.2. Optimization Design of E-REV. According to the optimization design, the weight of the key parts can be reduced effectively by using the strong and light materials.

Suppose the optimization target $P = \{P_1 \ P_2 \ \cdots \ P_m\}$, alternatives $A = \{A_1 \ A_2 \ \cdots \ A_n\}$, weight of every subgoal $W = \{w_1 \ w_2 \ \cdots \ w_n\}^T$, and multitarget optimization target matrix and the standardization matrix are defined as

$$a = \begin{array}{c} A_1 \\ A_2 \\ \vdots \\ A_n \end{array} \begin{bmatrix} a_{11} & a_{12} & \cdots & a_{1m} \\ a_{21} & a_{22} & \cdots & a_{2m} \\ \vdots & \vdots & \vdots & \vdots \\ a_{n1} & a_{n2} & \cdots & a_{nn} \end{bmatrix} \begin{bmatrix} w_1 \\ w_2 \\ \vdots \\ w_m \end{bmatrix},$$



(a) Deformation of battery pack 1 (b) Deformation of battery pack 2

Figure 4: The rear-end collision simulation of battery pack 1 and battery pack 2.

$$C = \begin{array}{c} A_1 \\ A_2 \\ \vdots \\ A_n \end{array} \begin{bmatrix} c_{11} & c_{12} & \cdots & c_{1m} \\ c_{21} & c_{22} & \cdots & c_{2m} \\ \vdots & \vdots & \vdots & \vdots \\ c_{n1} & c_{n2} & \cdots & c_{nn} \end{bmatrix} \begin{bmatrix} w_1 \\ w_2 \\ \vdots \\ w_m \end{bmatrix}, \quad (1)$$

where C is the standardization target matrix after dimension magnitude target value is standardized, $c_{ij} = a_{ij}/\sqrt{(\sum_{k=1}^{n} a_{ij}^2)}$, $i = 1, 2, \ldots, n$, $j = 1, 2, \ldots, m$, and a_{ij} is the parameter of A_{ij} under target value P_j. As the optimization value is as small as possible, the optimal value is defined as

$$F_j = \min c_{ij}. \quad (2)$$

The E-REV vehicle lightweight design target is set as P = {bending rigidity, torsional rigidity, plastic deformation, first bending mode, first torsional mode, weight, and parts price}. As optimization estimate algorithm, AHP method (AHP is the abbreviation of Analytic Hierarchy Process) is applied in optimization effective factors calculation of E-REV lightweight design. Comparison matrix according to 9-level score standard is described below:

$$P = \begin{bmatrix} 1 & \frac{1}{3} & \frac{1}{2} & \frac{1}{3} & \frac{1}{4} & \frac{1}{5} & \frac{1}{7} \\ 3 & 1 & 2 & 1 & \frac{1}{2} & \frac{1}{3} & \frac{1}{5} \\ 2 & \frac{1}{2} & 1 & \frac{1}{2} & \frac{1}{3} & \frac{1}{4} & \frac{1}{6} \\ 3 & 1 & 2 & 1 & \frac{1}{2} & \frac{1}{3} & \frac{1}{5} \\ 4 & 2 & 3 & 2 & 1 & \frac{1}{3} & \frac{1}{4} \\ 5 & 3 & 4 & 3 & 3 & 1 & \frac{1}{3} \\ 7 & 5 & 6 & 5 & 4 & 3 & 1 \end{bmatrix}. \quad (3)$$

The eigenvector matrix weighted values of different influence factors W_i = [plastic deformation, bending rigidity, torsional rigidity, first bending mode, first torsional mode, weight, and parts price] $= \sqrt[n]{\prod_{i=1,j=1}^{n} P_{ij}/(\sum_{k=1}^{n} P_k^2)}$ = [0.031, 0.083, 0.052, 0.083, 0.128, 0.218, 0.405]. The maximal eigenvalue of matrix is $\lambda_{\max} = \sum((PW)_i/W_i)/n = 7.213$. The consistency index CI $= (\lambda_{\max} - n)/(n-1) = 0.036$. By using the methods of table look-up we can know the consistency index RI = 1.32 and the consistency ratio CR = CI/RI < 1. Therefore, the effective factors proportion is rational.

The optimization function can be built considering every effective factor for E-REV lightweight design weighted values:

$$\begin{aligned} F(M_i) = \min \bigl[& F_1(M_i), F_2^{-1}(M_i), F_3^{-1}(M_i), \\ & F_4^{-1}(M_i), F_5^{-1}(M_i), \\ & F_6(M_i), F_7(M_i) \bigr]^T W_i^T, \end{aligned} \quad (4)$$

where $F_1(M_i)$ is plastic deformation and its calculation value is as small as possible. $F_2^{-1}(M_i)$, $F_3^{-1}(M_i)$, $F_4^{-1}(M_i)$, and $F_5^{-1}(M_i)$ represent the bending rigidity, the torsional rigidity, the first bending mode, and the first torsional mode. The higher their value is, the better the design is. So take its reciprocal. $F_6(M_i)$ and $F_7(M_i)$ are the weight and the part prices. Therefore the calculated value should be as small as possible.

The restriction condition can be derived:

$$\begin{gathered} F_1(M_i) < F_1(M_0), \\ F_i(M_i) > F_1(M_0); \quad i = 2, \ldots, 5, \\ F_6(M_0) \text{—} 80\,\text{kg} < F_6(M_i) < F_6(M_0); \\ F_7(M_0) \text{—} 30 \text{ thousand yuan} < F_7(M_i) < F_7(M_0), \end{gathered} \quad (5)$$

where $[F_1(M_0), \ldots, F_7(M_0)]^T$ are the optimization target default values. According to the prototype test data, the default values are [35 mm, 28000 N/mm, 19000 N·m/°, 45 Hz, 40 Hz, 1900 kg, ¥80,000]. The maximized reduction of the vehicle body weight and the part prices are 80 kg and ¥30,000, respectively (approximately $5,000). In order to reduce the calculating workload, the relationship between the whole part prices and the part weight is presented and solved by the linear curve.

Compared with the prototype vehicle, the front floor, the central floor, the roof cover, the threshold, and the central channel stiffener are not changed much, while the vehicle rear floor, the rear stringer, and the side hoarding have been strengthened. Compared with the prototype vehicle, the whole weight of E-REV before lightweight adds 5.206 kg. The weight of the rear floor and the rear beam has been increased about 2.5 kg. Based on the previous experiences, the affected parts are usually the BIW rear floor, the rear beam, the spare pool, and the rear stringer.

The weight of the parts is 0.5 kg when simplifying the optimization calculation process and accelerating the computation speed. The thicknesses of the 70 parts which are taken as the optimization targets are input the formula (4), and then calculate and gain the 4 optimal results by 91 iterations.

The weight of optimal point influential factors after standardization $W_0 = [0.051, 0.372, 0.042, 0.383, 0.115, 0.481, 0.324]$, while the optimal result $F_0(M) = $ [35.4 mm, 28146 N/mm, 18235 Nm/°, 43.6 Hz, 42.8 Hz, 1878 kg, ¥67,000].

According to the actual situation, the optimal results need to be adjusted again. However, according to the analysis and the calculating results, the part weight has a great influence on lightweight design and part price. Therefore, to control the part weight and the thickness is the most effective method for E-REV lightweight.

Based on the previous experiences, the two effective methods of reducing parts weight are to reduce the thickness of parts and to use high strength steel under the condition of keeping parts performance.

Suppose the thickness of steel is t and width is w; the face maximal stress of plate without damage is defined as

$$P_{\max} = wt\sigma_y, \tag{6}$$

where w and t are the width and thickness of plate and σ_y is the yield stress. If steel was changed by high strength steel, plate thickness with no plastic deformation is defined as

$$t' = \frac{t\sigma_y}{\sigma'_y}, \tag{7}$$

where σ'_y is high strength plate yield stress. Similarly, the high strength plate thickness with no plastic bending deformation is described as follows:

$$t' = \frac{t\sqrt{\sigma_y}}{\sqrt{\sigma'_y}}. \tag{8}$$

The manufacturability of complex parts during the lightweight design process needs to be considered. According to the simulation result, reset the weight optimization objective function to make the weight of optimized body structure as close as possible to the performance parameters optimization target, and then the optimization constraints are set. The parts rigidity and mode are the basic static and dynamic performance for traditional vehicle, therefore, setting the rigidity and mode of E-REV optimization constraints, building the weight optimization design program.

Table 4: The sensitivity of main parts weight and thickness.

Variable number	02	03	35	47	55	57
Contribution ratio (%)	17.3	21.1	1.23	0.91	0.87	0.63

The weight optimization target function and constraints are given by

$$\begin{gathered} F = \min W(X), \\ g_j(X) \le 0, \quad j = 1, 2, \ldots, m, \\ t_{i\min} \le t_i \le t_{i\max}, \quad i = 1, 2, \ldots, n, \end{gathered} \tag{9}$$

where F is target function, $W(X)$ is part weight function, $X = [t_1\ t_2\ \cdots\ t_n]^T$ is optimization vector, t_i is optimization design part thickness, $t_{i\min}$ and $t_{i\max}$ are optimization design upper limit and lower limit of part thickness, and $g_j(X)$ is constraint function.

Set the initial value of the optimization variable into the original thickness of the car body structure parts; X_0 is the initial optimization vector, and the iteration point is given by

$$\begin{gathered} X_{q+1} = X_q - q\frac{P\cdot\nabla W(X_q)}{\left\|P\cdot\nabla W(X_q)\right\|}, \\ P = I - G\left[G^T - G\right]^{-1} G^T, \\ G = \left\lfloor \nabla g_1(X_q)\ \ \nabla g_2(X_q)\ \ \Lambda\nabla g_m(X_q)\right\rfloor, \end{gathered} \tag{10}$$

where P is calculated factor, I is units matrix, and G is the gradient matrix of constraint function. The initial optimization can be gotten from part thickness and the optimization function iterates convergence along negative gradient direction. The optimal value would be gotten in calculated region. The sensitivity of main parts weight and thickness is shown in Table 4.

The number 02 and number 03 variables are the mass of battery pack 1 and battery pack 2, and number 35 variable is the mass of battery pack 1 mounting bracket, while number 47 variable is the thickness of the upper box. Number 55 and number 57 variables are the thickness of battery pack 2 lower box and cover plate. From the simulation results in Table 4, the weight sensitivity of battery pack 1 and battery pack 2 is the highest. It is obvious that reducing the weight of battery pack 1 and battery pack 2 is a great contribution to the lightweight design. The results of the optimizing calculation are similar to the simulation results. The total weight of battery pack 1 and battery pack 2 is 220 kg, which accounts for 81.5% proportion of the added weight of E-REV. Therefore, the lightweight of the battery packs and mounting bracket has the greatest influence on E-REV lightweight.

5. Analysis of E-REV Lightweight Design

5.1. The Lightweight Design of E-REV. Table 5 shows the bending sensitivity of main parts weight and thickness. Variable number 05 is the weight of charger on left rear

Table 5: The bending sensitivity of main parts weight and thickness.

Variable number	02	03	05	07	24	46
Bending rigidity contribution ratio (%)	1.7	2.5	0.25	0.48	0.24	0.03
Torsional rigidity contribution ratio (%)	0.9	1.1	0.11	0.09	0.07	0.02

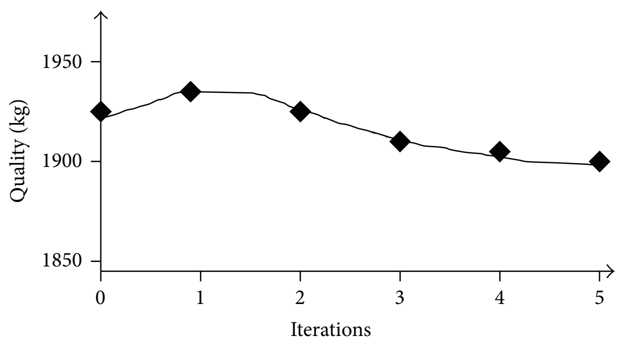


Figure 5: The simulation result of vehicle weight.

stringer, variable number 07 is the weight of drive motor on rear drive shaft, variable number 24 is the weight of lead-acid battery, and variable number 46 is the thickness of charger mounting bracket. According to the analysis results in Table 5, the weight and thickness of the main parts have little influence on bending sensitivity [14].

The parts' optimized thickness and the result of the optimized objective function are shown in Figure 5; the vehicle weight reduces from 1935 kg to 1861 kg. However, according to the actual steel plate standard, the optimized thickness should be adjusted again.

According to (8), the Q235 (yield stress is 215 MPa) steel sheet of the primary battery box is changed into the high strength steel sheet (yield stress is 410 MPa). The thickness $t = 2.0$ mm of Q235 sheet is reduced to the thickness $t' = 1.448$ mm of the new high strength steel sheet under the circumstance of the plastic deformation not occurring.

According to the simulation analysis, the weight loss results of main parts are shown in Table 6.

If the large capacity batteries are used, then total battery voltage drops from 352 V to 320 V and the total weight of the batteries reduces by 28.78 kg based on the constant total battery energy. In this case, the lightweight proportion is 13.08% and the lightweight effect is obvious.

In order to guarantee the strength and rigidity of the important loaded components during car body lightweighting design, the shape of some parts is changed from opening shape to closed shape. Then the thickness of those parts can be reduced and the strength and rigidity of those parts are greatly improved without added weight [15].

The lightweight of the fixed structure of the E-REV body is also designed. The thickness of the plate sheet reduces from 2.0 mm to 1.5 mm and the total weight of the car body reduces by 37.62 kg, the proportion of weight loss is 13.9%, and the total vehicle weight after lightweight is 1868.68 kg. The results agree well with the optimization function calculated results and the lightweight effect is obvious.

Table 6: Weight loss comparison of main parts in battery pack.

Parts	Thickness (mm)	Lightweight thickness (mm)	Reducing weight (kg)
Battery pack 1 upper box	2.0	1.0	2.28
Battery pack 1 low box	2.0	1.5	0.186
Battery pack 1 mounting bracket	2.0	1.5	0.22
Battery pack 2 cover	2.0	1.0	5.26
Battery pack 2 low box	2.0	1.0	7.26

Table 7: The rear-end collision simulation comparison of E-REV before and after lightweighting the main parts.

	Before lightweight (mm)	After lightweight (mm)	Difference (%)
Maximal deformation of vehicle body	350	344	1.7
Maximal deformation of battery pack 1	31	22	29
Maximal deformation of battery pack 1 mounting bracket	14.1	13.8	1.4
Maximal deformation of battery pack 2	35	27.8	20.6
Maximal deformation of battery pack 2 mounting bracket	26.7	25.3	3.1

5.2. The Rear-End Collision Simulation of E-REV after Lightweight. The lightweight FEM model of E-REV is input again and is calculated. From the simulation result, the mounting bracket of battery pack 1 is impacted and the deformation process and the time of appearing maximal acceleration are similar to before. It explains that energy absorption capability of battery pack 1 mounting bracket does not drop after lightweight design. The crash force is close to the crash force before lightweight under rear-end collision test. The rear-end collision simulation comparison of E-REV before and after weight loss of main parts is shown in Table 7.

Table 8: The mode simulation of E-REV before and after lightweighting the main parts.

	Bending mode (Hz)	Torsional mode (Hz)
Before lightweight	44.5	38.3
After lightweight	43.7	42.4

To some extent, the maximal deformations of two battery packs' mounting brackets drop and the box structures are strengthened after the lightweight design as shown in Table 7. The collision protection of the battery packs has strengthened and then the electric shock risk will be reduced because the risk of the battery module damage and electrolyte leakage is largely reduced. The results are very important for passing standards GB 20072 and GB/T 19751 during the rear-end collision test of E-REV.

The mode simulation results of E-REV are shown in Table 8. The simulation results are based on the complete BIW (body in white) with dormer, front subframe, and front and rear windshield. They show that the first-order bending mode of E-REV after lightweight is close to the result before lightweight and the torsional modal frequency greatly improved. Similarly, they can predict that rigidity of E-REV BIW after lightweight is close to the result before lightweight.

As the lightweight model has passed rear-end collision test, it can be predicted that the E-REV after lightweight will get better experimental results and pass the rear-end collision test.

6. The Rear-End Collision Test of E-REV after Lightweight

The rear-end collision test of lightweight E-REV operates according to standard GB 20072-2006. The collision speed is 50 ± 2 km/h and fuel is replaced by water before the test.

The collision acceleration curves of battery pack 1 bracket under rear-end collision test before and after weight loss of main parts are shown in Figure 6. Compared with the simulation, the peak acceleration and appearing time have some deviation, but the acceleration curves are coincident. All the curve trends are similar and the discrepancy is very little (<10%) at the maximal acceleration happen time; then the test data show that the simulation model is accurate and can be used to analyze the rear-end collision safety of E-REV.

Figure 7 is the knocked-down picture of E-REV battery pack 1 and battery pack 2 under rear-end collision test after weight loss of main parts. As shown in Figure 7(a), the battery mounting bracket deforms, but the deformation is not large and the battery module of the battery pack 1 is not obviously damaged. So the battery mounting bracket plays an important role in resisting deformation and energy absorption. After rear-end collision test, the left-front module bracket of the battery pack 2 has a little deformation, but bolt, cable, harness, and connection are not damaged. The bakelite plate is broken but with no battery damage and leak, as shown in Figure 7(b). It is found that the voltages of three battery modules are abnormal in battery pack 2, but the voltages of other battery modules in battery pack 1 and battery pack 2 are normal.

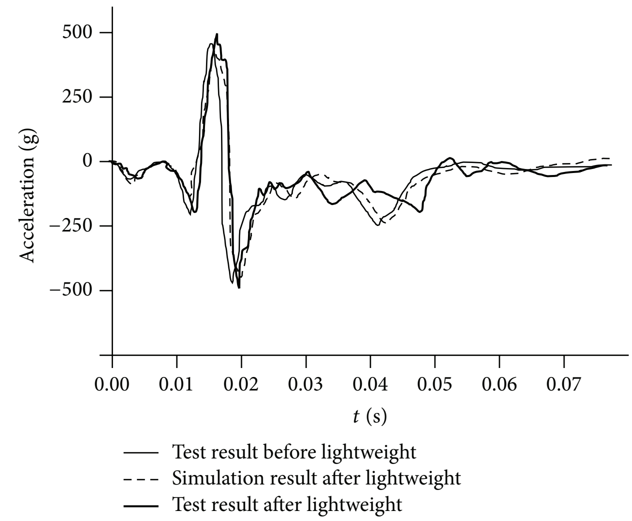


Figure 6: The collision acceleration of battery pack 1 bracket under rear-end collision test before and after lightweighting the main parts.

Table 9: The test result of E-REV vehicle modality after lightweighting the main parts.

	Test	Simulation	Deviation
Torsional mode (Hz)	41.3	42.4	2.66%
Bending mode (Hz)	42.9 Hz	43.7	1.86%

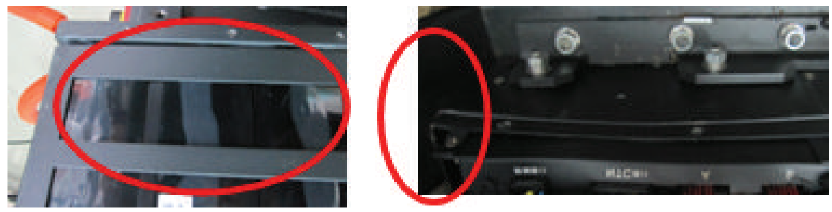
The experimental results can pass the requirement of standard GB/T 19751.

The simulation results of the maximal vibration displacement under rear-end collision test after lightweighting the main parts can be shown in Figure 8. The locations of the maximal vibration displacement and bending displacement are consistent with the test results. The rear-crash test method is done according to the test process in GB 20072.

The test results of E-REV vehicle modality after weight loss of main parts are given in Table 9. They draw a conclusion that they are similar to simulation result of torsional and bending modal after lightweight. The reliability of simulation model and theory analysis is proved to be right from data in Table 9. The BIW mode of E-REV after weight loss of main parts is similar to before.

The deviation of battery pack 1 maximal deformation simulation compared with test data reaches to 15.7%, but other parts' maximal deformation simulation deviation is accurate and less than 10%. On the other hand, it demonstrates that the simulation model is reliable. The comparison data are given in Table 10.

The rear-end collision test of E-REV completely satisfied standards GB/T 19751-2005 (hybrid electric vehicles safety specification) and GB 20072-2006 (the requirements of fuel system safety in the event of rear-end collision for passenger car) [16, 17].



(a) Deformation of battery pack 1 (b) Deformation of battery pack 2

FIGURE 7: The knocked-down picture of E-REV battery pack 1 and battery pack 2 under rear-end collision test after lightweighting the main parts.

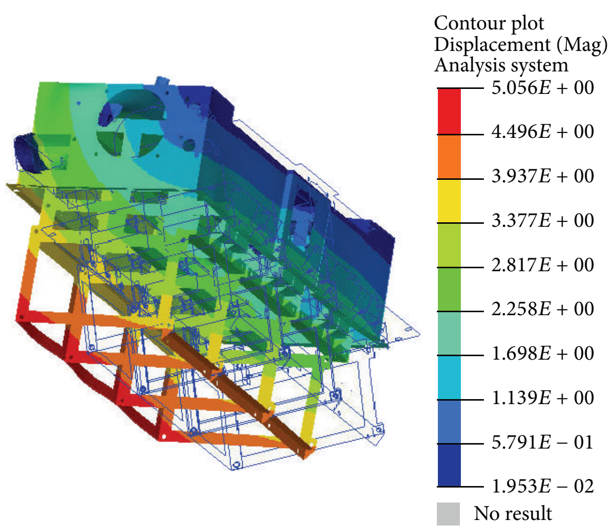


FIGURE 8: The deformation comparison of simulation picture of E-REV battery pack 1 inner structure under rear-end collision test after lightweighting the main parts.

TABLE 10: The rear-end collision test of E-REV after lightweighting the main parts.

	Test	Simulation	Deviation
Maximal deformation of vehicle body (mm)	341	344	1.7%
Maximal deformation of battery pack 1 (mm)	19	22	15.7%
Maximal deformation of battery pack 1 mounting bracket (mm)	14	13.8	1.43%
Maximal deformation of battery pack 2 (mm)	28	27.8	0.7%
Maximal deformation of battery pack 2 mounting bracket (mm)	24	25.3	5.4%

7. Conclusions

The total mass of E-REV is heavier than the prototype vehicle. In view of the massive influence of the total mass of E-REV on fuel economy and passive safety, the lightweight design of E-REV is analyzed and the optimized objective function is built based on the rear-end collision safety. The influence factor and weight sensitivity of E-REV lightweight design are analyzed. According to the simulation results, the battery pack structures and the battery pack mounting brackets are mainly lightweight objects to be optimized to drop E-REV weight. It is proved that the total mass of E-REV is cut down largely under the precondition of enough passive safety. So the fuel economy and emission of the vehicle are improved largely after lightweight. It is proved that E-REV has enough safety by rational lightweight optimization methods from the rear-end collision results of the simulation and test. It can effectively reduce the weight of key parts using the high strength and light materials under the precondition of crash safety and manufacturing cost control by the optimization design. The multitarget optimization function is an important method for lightweight. The results show that the lightweight design method is effective and the simulation model is correct. It is possible to analyze other similar lightweight problems of new energy vehicles by the similar simulation model and optimization methods.

Conflict of Interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

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